

**STUDY OF PLATE-FIN HEAT EXCHANGER AND COLD PLATE FOR THE
ACTIVE THERMAL CONTROL SYSTEM OF SPACE STATION**

Final Report

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ABSTRACT

Plate-fin heat exchangers will be employed in the Active Thermal Control System of Space Station Freedom. During ground testing of prototypic heat exchangers, certain anomalous behaviors have been observed. Diagnosis has been conducted to determine the cause of the observed behaviors, including a scrutiny of temperature, pressure, and flow rate test data, and verification calculations based on such data and more data collected during the ambient and thermal/vacuum tests participated by the author. The test data of a plate-fin cold plate have been also analyzed. Recommendation was made with regard to further tests providing more useful information of the cold plate performance.

INTRODUCTION

The compactness and high efficiency of plate-fin heat exchangers have led to their application in the Active Thermal Control System of the Space Station Freedom. In such application, ammonia is evaporated by means of warm water flowing through the heat exchanger composed of a number of finned flow passages. In a similar device, cold plate, the heat applied to a plate is removed by the evaporating ammonia flowing in a finned passage on the other side of the plate. Both plate-fin heat exchanger and plate-fin cold plate have been tested in the Space Station Ground Test Article (GTA) at NASA Johnson Space Center.

PLATE-FIN HEAT EXCHANGER

Introduction

The prototypic plate-fin heat exchanger (PPFHX) tested in GTA was manufactured by Allied Signal Aerospace Company, with a core of 10.50 in \times 2.75 in \times 1.65 in, which is composed of 10 water passages interlaminated with 9 ammonia passages with all passages featuring offset fin structure. The results of the February 1992 GTA test indicated a performance inferior to the design expectation. The objectives of the present work are (a) to diagnose the PPFHX based on the existing test data, (b) to design and conduct further tests, and (c) to characterize the heat exchanger based on all test data available.

Diagnostic Analysis

During the GTA test conducted in February 1992, very large pressure drop has been observed on the water side (about 4.7 psid compared with 1.1 psid design specification at 1800 lbm/hr). Such high pressure drop is considered due to blockage of flow passages. The flow passages can be blocked if the heat exchanger was not properly brazed during manufacturing. According to the manufacturer, the plates and fins were brazed together by interposed braze sheets, which melted and being sucked into the interstices during a controlled oven-heating process. If the heating process is not well controlled in terms of temperature levels, timing, and uniformity of heat, stray molten braze may partially or even entirely block the finned passages. The current heat exchanger is a custom-built unit which required a special heating process to braze. However, according to the manufacturer, the unit was heated along with other units following a regular heating procedure because of constraint of time and cost.

An ensuing problem of blockage of water flow passages is uneven heating of ammonia flow passages. In a blocked water passage, the flow rate is low, and the heat transfer rates to the adjacent ammonia passages are low. On the other hand, the water flow rate in an unblocked (or less blocked) passage is high, and the heat transfer rates to the adjacent ammonia passages are high. The heat transfer rate can be so high that the ammonia becomes superheated vapor at the outlet. The superheated vapor then mixes at the ammonia outlet with the saturated vapor and saturated liquid from passages with low heat rates. The saturated liquid temperature ought to be detected if a thermocouple is installed at the bottom of the ammonia outlet manifold. The saturated liquid can be vaporized by the superheated vapor. However, the vaporization rate is low because of the low heat transfer coefficient between the superheated vapor and the saturated liquid.

Therefore, it takes a long time for such two-phase flow to reach a thermal equilibrium. The temperature measurement taken at the ammonia outlet 10 in away from the body of the heat exchanger indicated significant superheating within the heat exchanger.

Superheating of ammonia results in a low heat exchanger performance due to the low heat transfer coefficient between the ammonia vapor and the fin structure which is supposed to be wetted by ammonia liquid. This is considered the reason why the present heat exchanger did not measure up the design expectation.

The core of the heat exchanger was examined using a borescope in order to detect any visible blockage. There was no sign of blockage in the fin structure as viewed from the water inlet and outlet manifolds. The size of the borescope prohibited viewing of the internal core structure.

Another concern regarding the inferior heat exchanger performance is the material compatibility. The braze alloy used in binding the plates and fins, AMS 4778, contains 91% of Ni, 4.5% of Si, 3.5% of B, 0.8% of Fe, and 0.05% of C, of which iron has "severe effect" when in contact with ammonia (Grainger Catalog No. 380). The supplier of the heat exchanger indicated that no compatibility test had ever been conducted with such braze alloy and ammonia, even though a report on the ammonia compatibility tests of a number of other braze alloys was provided.

The heat exchanger has been returned to the manufacturer after the GTA test in June, 1992, to go through regular testing which should have been conducted before shipping.

Experimental Results and Discussion

After reviewing the February 1992 test data, the author participated in the GTA test in June. The results of the June test, both ambient and thermal/vacuum, are presented and compared with February 1992 test data (Sifuentes et al., 1992) below. All the data were taken at the water flow rates between 1781 and 1837 lbm/hr, and the ammonia flow rates ranging from 75 to 113 lbm/hr. The heat load or the heat exchange rate between two streams, Q , the quality at the ammonia outlet, x , the heat exchanger effectiveness, ϵ , and overall heat transfer coefficient, UA , are calculated using the following equations:

$$Q = \dot{m}_{H_2O} \times C_{p,H_2O} \times (T_{H_2O,in} - T_{H_2O,out})$$

$$x = \frac{Q}{\dot{m}_{NH_3} h_{fg,NH_3}}$$

$$\epsilon = \frac{T_{H_2O,in} - T_{H_2O,out}}{T_{H_2O,in} - T_{NH_3,in}}$$

$$UA = \frac{Q}{\Delta T_{lm}}$$

where ΔT_{lm} is the log-mean temperature difference. Figure 1 shows the expected trend of heat load increasing with approach temperature, the temperature difference at the inlet. There is a reasonable agreement among the data of three different tests, viz., February ambient test, June ambient test, and June thermal/vacuum test. However, the design

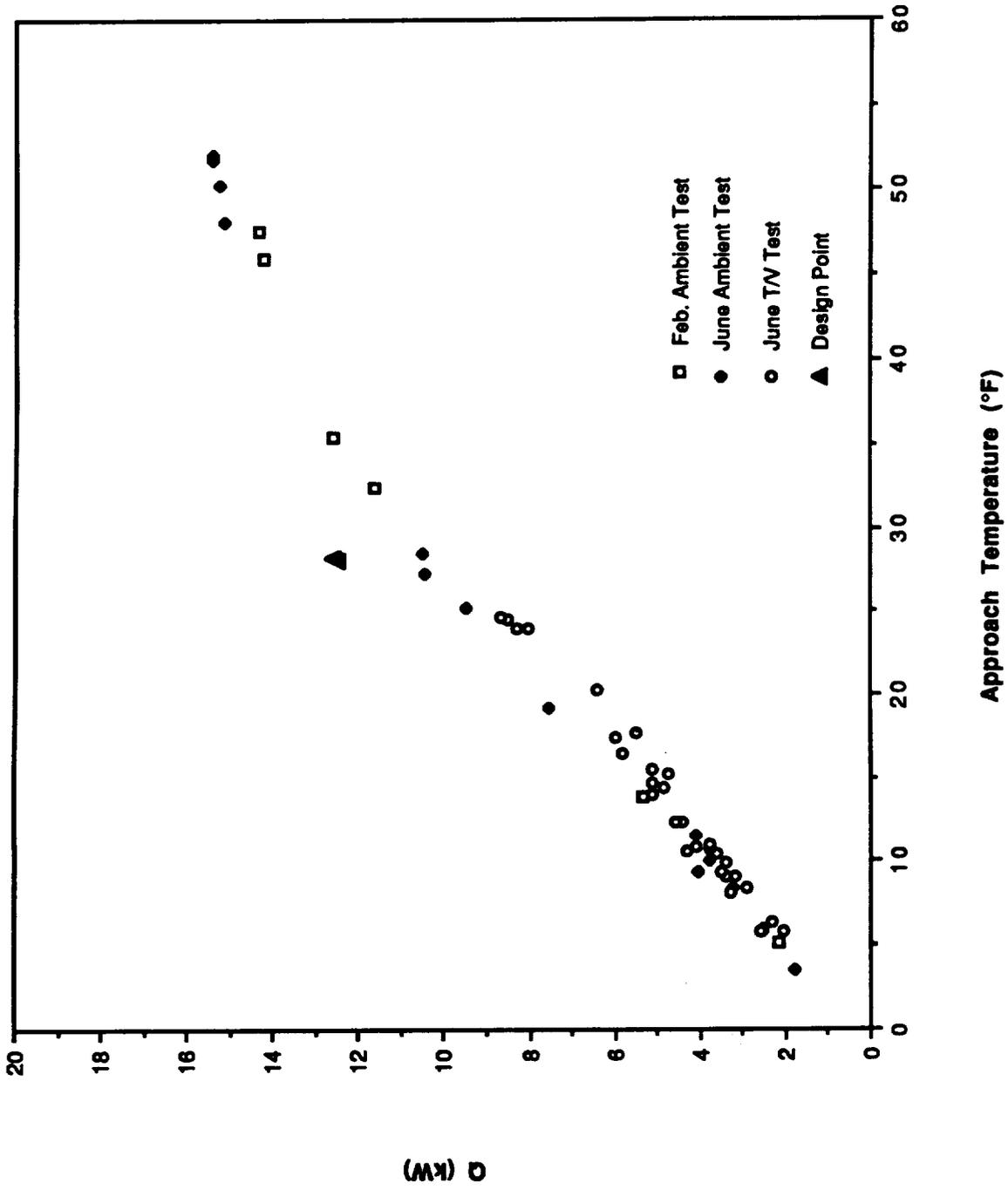


Figure 1. - Heat Load vs Approach Temperature

points are above the data curve, indicating that at the design approach temperature (28°F), the heat exchanger falls short of achieving the desired heat transfer rate (12.5 kW) by about 16%. The pinch temperature (difference between the water outlet and the ammonia inlet temperatures) versus approach temperature data exhibited in Fig. 2 show that all test data demonstrate pinch temperature increasing with approach temperature. It is also indicated that at the design approach temperature, the pinch temperature is about three times as large as the design specification (3°F). The pinch temperature vs. heat load plot (Fig. 3) shows that at the design heat load (12.5 kW), the pinch temperature is approximately four times as large as the design specification. Since quality is directly proportional to heat load, Fig. 4 is simply a different presentation of Fig. 1.

Both the effectiveness and the overall heat transfer coefficient data presented in Figs. 5 and 6, respectively, demonstrate decreasing trends with heat load. This trend is due to a large dryout area in the ammonia passages at a high heat rate, resulting in a low heat transfer coefficient as well as heat exchanger effectiveness. Figure 7 in fact is a different presentation of Fig. 6; however, it clearly exhibits a trend contrary to all the published results that heat transfer coefficient increases with quality. Heat transfer coefficient increases with quality because at a high quality, the fin surfaces are covered with liquid films, and the heat transfer coefficient of thin film evaporation increases with smaller film thickness as quality increases. The reverse trend of the present data is due to the large dryout area in the heat exchanger as quality increases with heat load.

Conclusion of Experimental Study

All the data obtained under different times and test conditions demonstrated reasonable agreement and consistent trends, and strongly suggesting a heat exchanger performance inferior to the design specification. The effectiveness and overall heat transfer coefficient data support the proposed diagnostic mechanism for the inferior heat exchanger performance of premature ammonia side dryout due to uneven water side heating.

Evaluation of Proposed Heat Exchangers for Prototypic Test Article (PTA)

Ten heat exchangers have been ordered from Hughes-Treitler Mfg. Corp. for the forthcoming Prototypic Test Article (PTA). These are also offset plate-fin heat exchangers with slightly different design specifications than the Allied Signal GTA heat exchanger. The proposed design was evaluated. The major difference compared with the GTA heat exchanger seems to be in the flow configuration. The GTA heat exchanger operates in parallel flow, while the PTA heat exchanger will operate in counterflow arrangement. The possible advantage of a parallel-flow evaporator is the large temperature difference at the inlet which facilitates inception of nucleate boiling on the cold (ammonia) side. However, such advantage does not exclude the possibility that a well-designed counterflow unit can do equally well. In fact, it has been observed that nucleate boiling does not occur in the evaporating flow in an offset-finned passage (Carey and Mandrusiak, 1986, Mandrusiak et al., 1988). Therefore, the large inlet temperature difference in a parallel-flow evaporator perhaps does not substantiate any advantage over a counterflow evaporator.

PLATE-FIN COLD PLATE

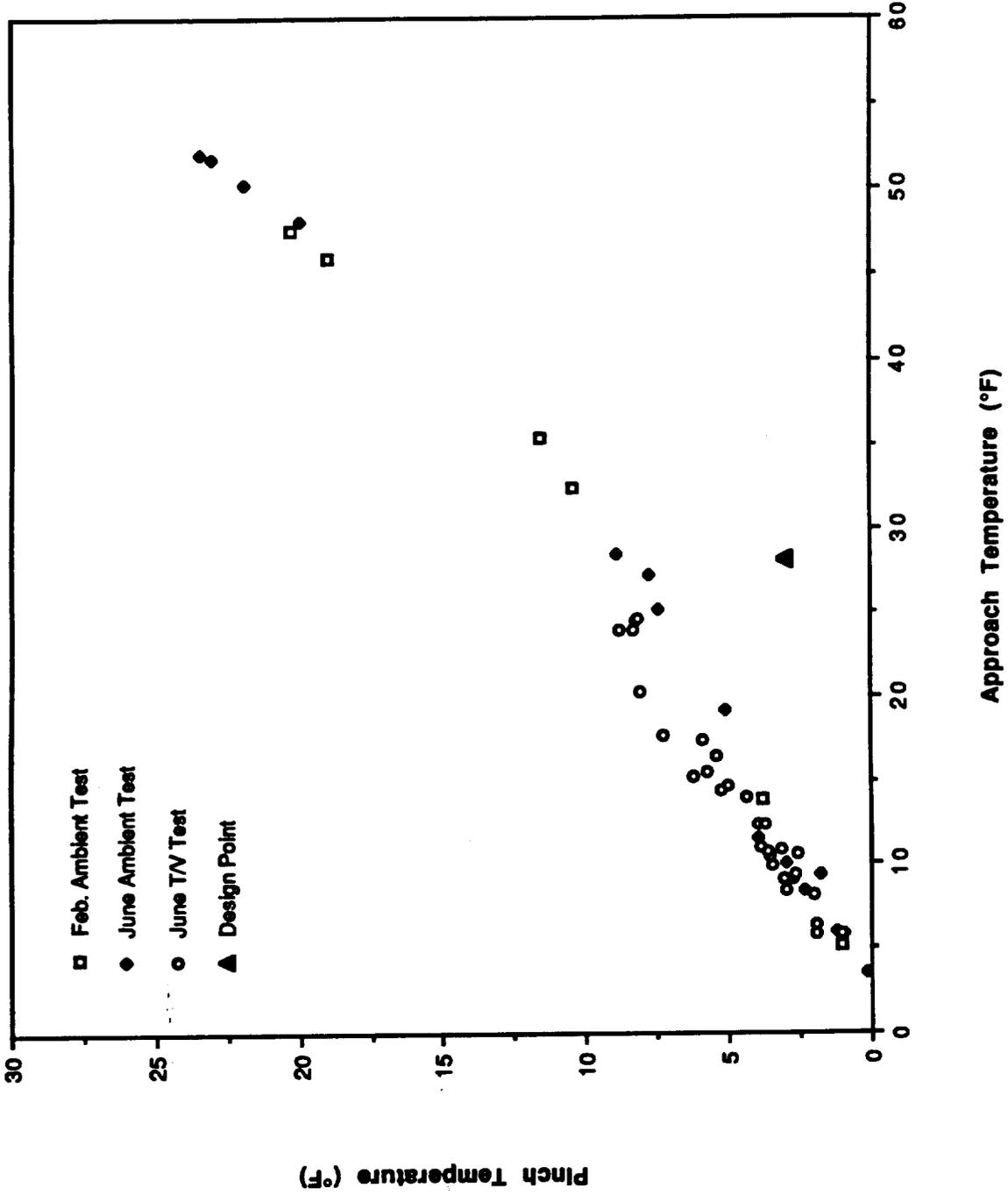


Figure 2. - Pinch Temperature vs Approach Temperature

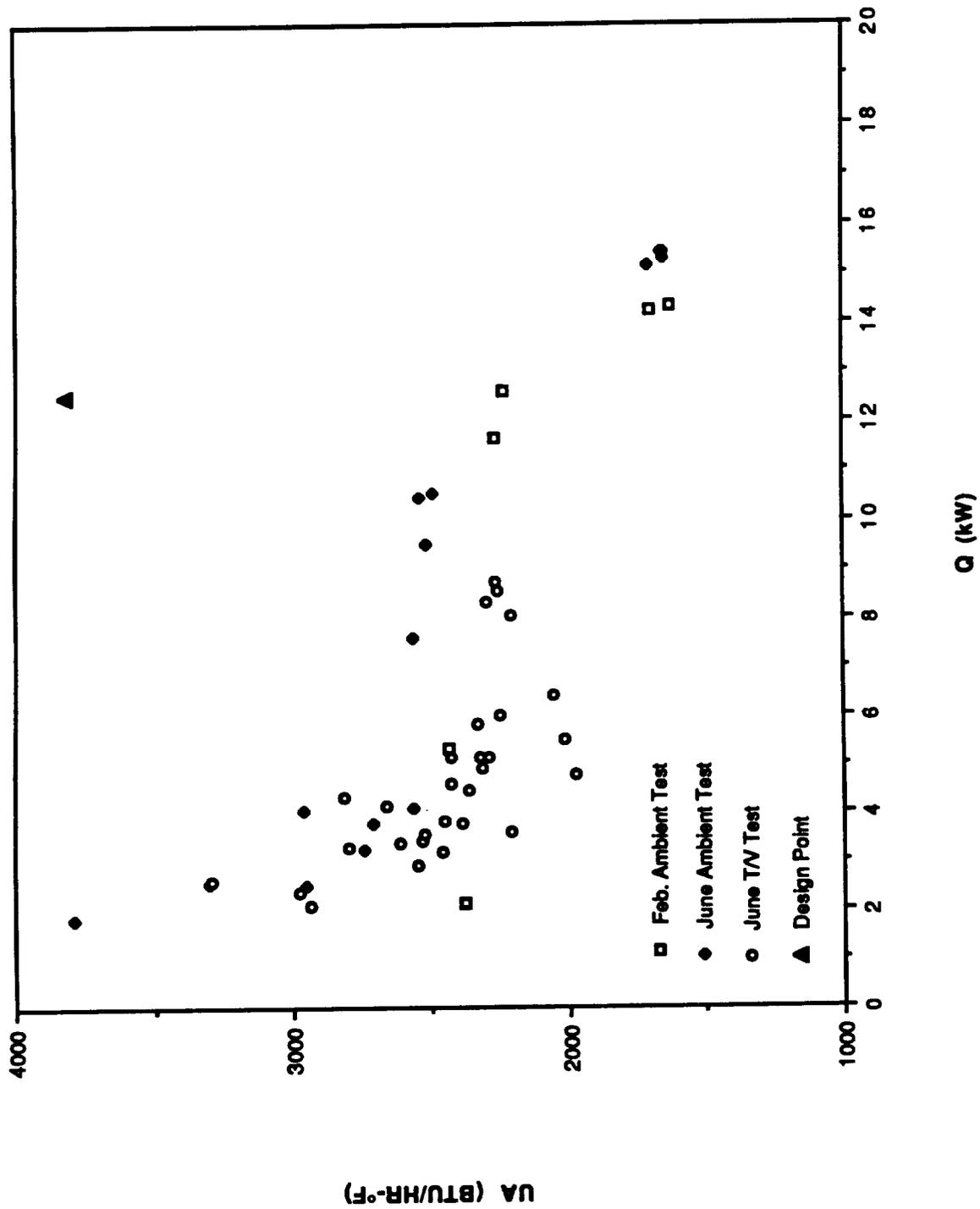


Figure 6. - Overall Heat Transfer Coefficient vs Heat Load

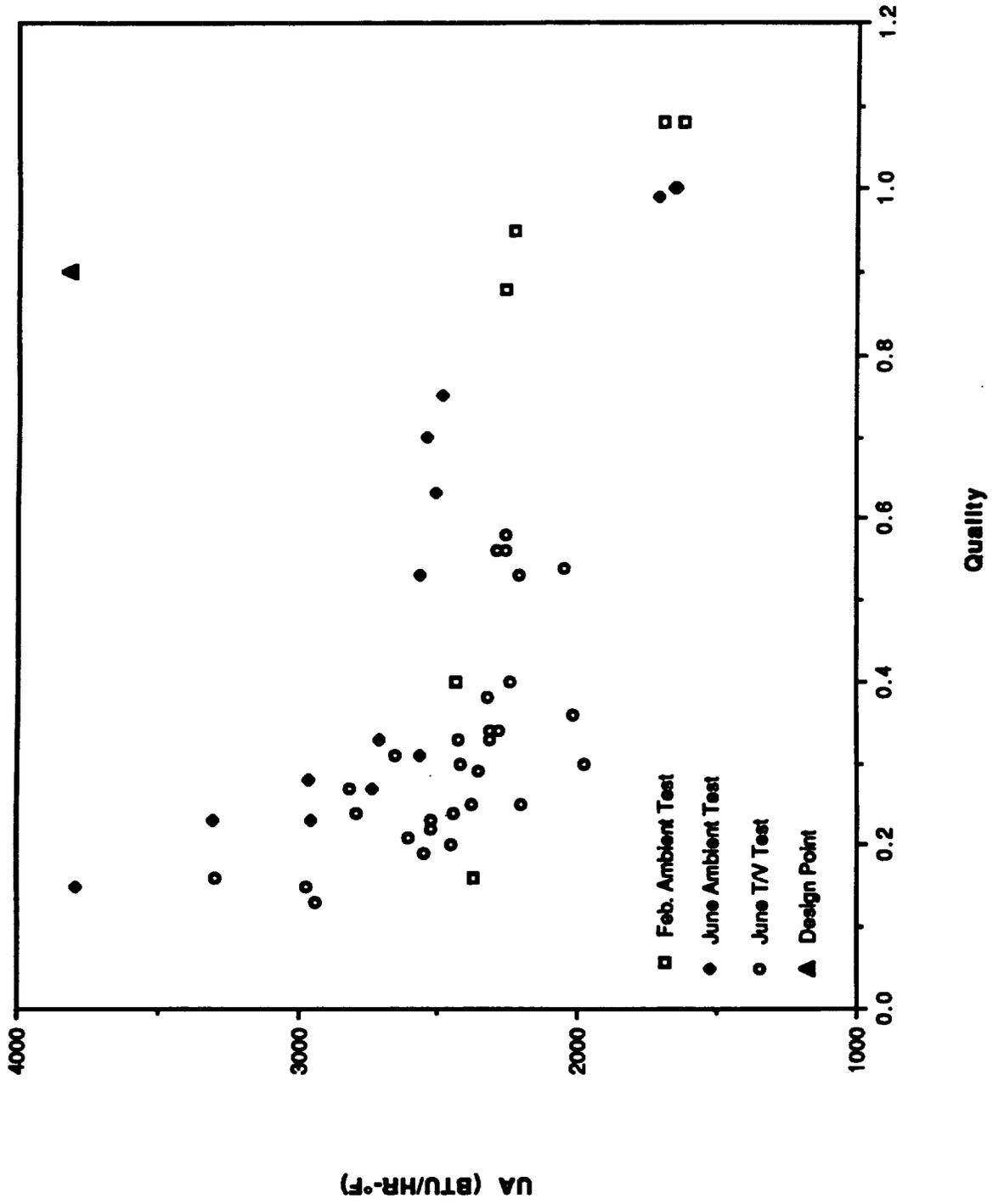


Figure 7. - Overall Heat Transfer Coefficient vs Quality

Introduction

An offset fin cold plate was tested in the GTA for its appropriateness as a means of temperature control of thermally active units such as electronic components. The Allied Signal plate-fin cold plate (26.90 in \times 26.65 in \times 0.85 in) has one serpentine offset-finned flow passage sandwiched between two plates, as shown in Fig. 8. Both the fins and the plates are of commercial 6061 aluminum. Commercial electric heaters (Minco etched-foil heaters) are installed on the top plate outside surface to simulate the heat sources. Tape thermocouples are installed on the bottom plate outside surface. The objective of this part of study is (a) to quantitatively characterize the performance of the cold plate by calculating the heat transfer coefficient, (b) to compare the performance data with published results, and (c) to develop improved performance prediction method for cold plates of this type.

Analysis

Considering one dimensional heat transfer across the cold plate, the temperature profile is qualitatively shown in Fig. 8. The broken curve across the finned flow passage indicates the unknown temperature profile along the fin, depending on the convective heat transfer coefficient on the fin surface. Owing to the insulation at the bottom plate surface, the temperature distribution across the bottom plate should be very uniform. Therefore, the temperature at the bottom plate surface, as measured by the thermocouples, is very close to the bottom wall temperature of the finned flow passage. The cold plate data collected in February 1992 test generally demonstrated higher wall temperatures at the inlet and lower wall temperatures at the outlet, a trend consistent with the published results that heat transfer coefficient increases with quality.

However, the present temperature data at the bottom plate surface are not appropriate for the calculation of the heat transfer coefficient of the cold plate. Since only the top plate was heated and the bottom plate was insulated, the heat transfer coefficient should be based on the top wall temperature which, however, was not measured in the test. Even if heat transfer coefficient is determined based on the bottom plate surface temperature, there is no published experimental data or correlation available for comparison. The published works on offset fin heat transfer data are mainly for heat exchangers with the finned flow passage subject to heating on both walls forming the passage (e.g., Chen et al., 1981, Robertson and Lovegrove, 1983, Panchal, 1989, Panchal and Arman, 1991). The only data available for the condition of one wall heated and the other wall insulated, as in the case of the present cold plate, are for water and methanol (Mandrusiak, et al., 1988) and *n*-butanol (Carey and Mandrusiak, 1986). No study has been conducted for the heat transfer of an evaporative ammonia flow in a passage with offset fins and with only one wall heated. A proposal has been submitted to NASA/JSC, which includes such experiment to be performed at Texas Tech University.

Conclusion

In summary, the results of the present study on plate-fin cold plate suggest the following: (a) The data collected for the present cold plate are not appropriate for the development of a prediction correlation. (b) A reliable prediction correlation for the cold

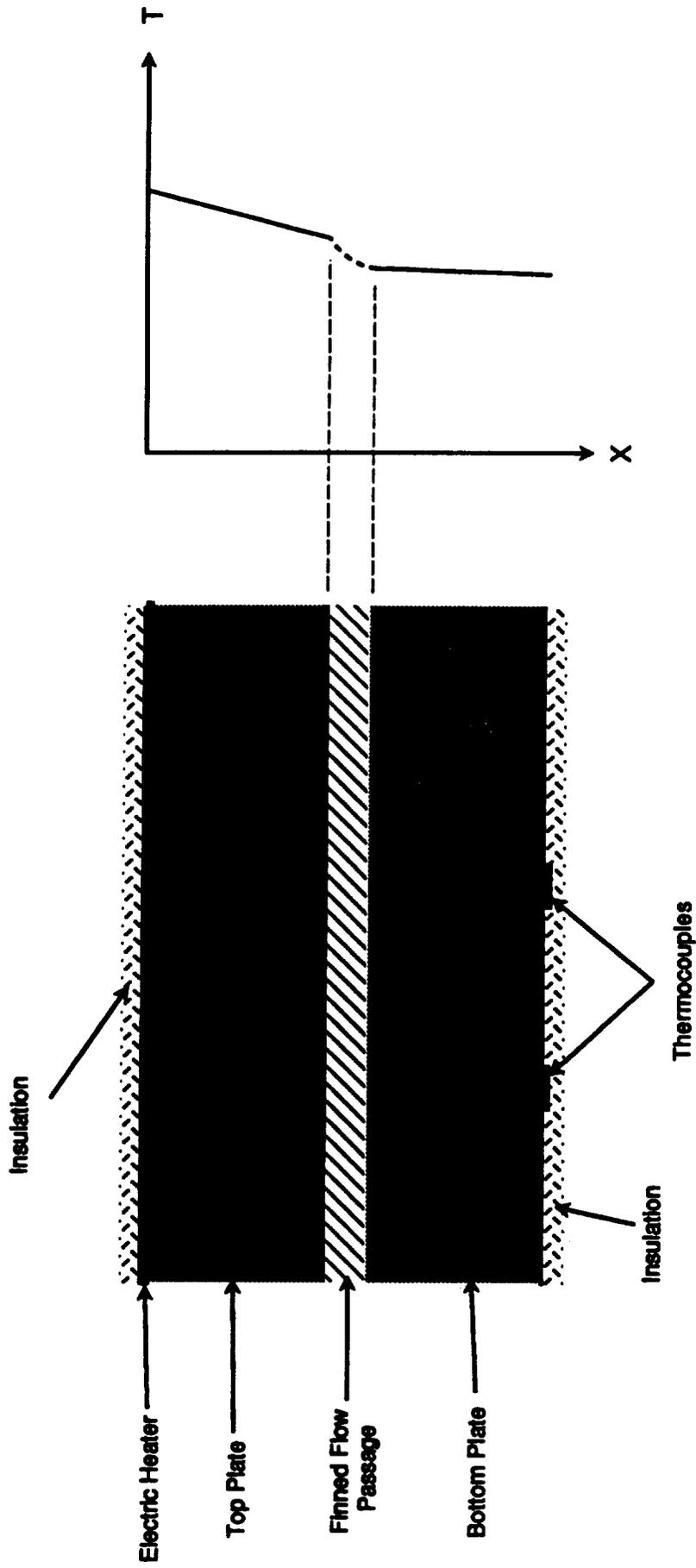


Figure 8. - Plate-Fin Cold Plate

plate should be based on the temperature of the heated wall and should be independent of the plate thickness. A design practitioner can predict the operating temperature of the heat source mounted on the heated plate based on such correlation and a conduction analysis across the plate. (c) There is a need of research for the evaporation of ammonia in a plate-fin cold plate with only one wall heated to provide heat transfer data for the development of reliable correlations. It is recommended that the heated wall temperature of the finned flow passage be determined by extrapolation of the temperature profile based on the readings of three or more thermocouples installed at different locations across the thickness of the heated plate.

REFERENCES

Carey, V. P., and Mandrusiak, G. D., 1986. "Annular Film-Flow Boiling of Liquids in a Partially Heated, Vertical Channel With Offset Strip Fins," *Int. J. Heat Mass Transfer*, Vol. 29, No. 6, pp. 927-939.

Chen, C. C., Loh, J. V., and Westwater, J. W., 1981. "Prediction of Boiling Heat Transfer Duty in a Compact Plate-Fin Heat Exchanger Using the Improved Local Assumption," *Int. J. Heat Mass Transfer*, Vol. 24, No. 12, pp. 1907-1912.

Grainger Catalog No. 380, W. W. Grainger, Inc., pp. 2251.

Mandrusiak, G. D., Carey, V. P., Xu, X., 1988. "An Experimental Study of Convective Boiling in a Partially Heated Horizontal Channel With Offset Strip Fins," *J. Heat Transfer*, Vol. 110, pp. 229-236.

Panchal, C. B., and Arman, B., 1991. "Analysis of Condensation and Evaporation of Ammonia/Water Mixtures in Matrix Heat Exchangers," paper presented at 1991 ASME/AIChE National Heat Transfer Conference, Minneapolis.

Panchal, C. B., 1989. "Analysis of Flow Boiling of Ammonia and R-114 in a Matrix Heat Exchanger," Proc. 1989 ASME/AIChE National Heat Transfer Conference, Philadelphia.

Robertson, J. M., and Lovegrove, P. C., 1983. "Boiling Heat Transfer With Freon 11 (R 11) in Brazed Aluminum, Plate-Fin Heat Exchangers," *J. of Heat Transfer*, Vol. 105, pp. 605-610.

Sifuentes, R., Bryant, M., Teunisse, B., Lee, L., 1992. "GTA Phase II Ambient Test Quick-Look Summary," March 18, 1992.

